Description

CAM DRIVE MECHANISM

[1] This invention solves the known problem of the desmodromic, or positive, control of a reciprocating member comprising a pair of rollers riding on a uni-lobe cam.

[2] By uni-lobe cam is meant a cam whose one rotation corresponds to only one reciprocation of the reciprocating member.

[3] By desmodromic or positive control is meant that if the one roller is in contact with the cam, the other roller is no more than the running tolerance away from the cam.

[4] In the art there are drive mechanisms using multi-lobe cams, as in US 4,545,336 patent or uni-lobe cams as in GB 891,490.

[5] The uni-lobe is simpler, smaller for the same stroke and can use counterweights, fixed on its shaft, for balancing inertia loads.

[6] It is possible for a cam mechanism to employ a second camshaft to bear the thrust loads, or can bear the thrust loads on walls, immovable or rotatable to provide variable compression.

[7] The latter needs not a gearing of high strength/high accuracy/small clearance to connect the cooperating camshafts, is cheaper to make and compact.

[8] The uni-lobe cam must control desmodromicly the reciprocating member in both directions. Unless the necessary cam profile can be defined as strictly as to provide the demanded quality and accuracy of the reciprocation the control is impossible, the cam cannot meet the contemporary demands.

[9] As a roller rolls along a cam, it contacts the cam along the cam profile but the center of the roller travels along its own path which, being the trace of the center, can be called the centers curve.

[10] The centers curve derives from the cam profile and the radius of the roller. Likewise, keeping the center of the roller on the centers curve and moving it along its successive positions, the roller defines the profiles of two cams, one external and one internal. If the axis of a milling cutter follows the centers curve, while removing material from a plate, it is machining the two cams mentioned.

[11] Any point on the centers curve derives the respective point of the cam profile by drawing a line, of length equal to the radius R of the roller, perpedicular to the centers curve at this point, as Fig. 38 shows.

[12] In GB 891,490 patent the control of the reciprocation is provided by a uni-lobe cam whose geometry is described in Page 3, lines 50 to 69, and in Fig. 5. In GB 891,490 the profile is defined as the union of two concentric circular arcs and two connecting curves such that the length of the line, defined by the contact point of the top roller and the contact point of the bottom roller, is constant. In the present invention only the

distance between the centers of the rollers is constant; the distance between the contact points of the two rollers is substantially variable, varying from a minimum length, equal to the distance of the centers minus the sum of the radiuses of the two rollers, to a maximum which, depending on the roller size, may become more than 10% bigger than the minimum for reasonable size of roller. Also, in GB 891,490, although the two rollers ride on a common cam, it appears that the top roller diameter is larger, probably because of its heavier loads. In the present application the mathematical derivation of the cam profile makes it clear that it is impossible to control the piston desmodromicly with a common cam and rollers of different diameter.

[13] The constant breadth cam of US 0020043225 patent application can control the system desmodromicly only if the diameter of the rollers is equal to zero, for reasonable diameter of the rollers the mechanism cannot be desmodromic.

[14] Besides its main cam lobe, the engine of the US 4,493,296, Fig. 22, needs a secondary groove-cam, i.e. internal cam, and additional secondary rollers for the restoring of the piston, while the opposed pistons move independently. If, as Fig. 20 of the present invention shows, the profile of the main cam is properly derived then the groove and the secondary rollers are unnecessary and the pairs of the opposed pistons can be united in a single body reducing inertia loads and friction. The typical groove cam used in US 4,493,296 and many other patents of the prior art, can be replaced by a pair of complementary external cams on the same camshaft, as shown in Fig. 47 to 49. Compared to groove cams, the use of a camshaft comprising a first external cam, to constrain the reciprocation at one direction, and a second complementary external cam, to constrain the reciprocation at the opposite direction, provides more robust cams of reduced size, easier construction etc. In Fig 47 and 48 the secondary external cam is selected to be a circle, while in Fig. 49 the general case is illustrated. The only requirement for the centers curves E1 and E2 is AC=BD=constant, i.e. given the CP1 cam profile or the piston's position function Y(f), the radiuses R1 and R2 and the constant distance of their centers, the centers curve E1 derives, then the E2 from the E1 and finally the complementary cam profile CP2.

[15] The version shown in Fig. 19, 26 and 27 necessitates a second camshaft but eliminates the second roller cam follower assembly. As long as the top cam is free of combustion loads, its size can be reduced in order to reduce the piston rod height and mass.

[16] In a preferred embodiment the motion converting mechanism of Fig. 8 consists of a pair of counter-rotating shafts (7) and (8).

[17] The shaft (7) has double-disk cams (9) and (10) to allow room for the cam (11) of the shaft (8).

[18] The profile of the cams, i.e. the control surface of the cams is made so that to

derive a harmonic reciprocation for the piston rod.

[19] By the mathematical term harmonic it is meant a strictly sinusoidal motion versus the time, i.e. versus the shaft-angle in the case of a single-lobe cam and versus the shaft-angle times the lobe-number for the cases of multi-lobe cams.

[20] The balance of inertia forces and moments for such a reciprocation is simple, even for a single cylinder or a twin, by virtue of a couple of counterweight webs fixed on the shafts, but only in case of single-lobe cams.

[21] The additional merit of the three-in-line of the Fig. 8, as compared to the single or twin, is that besides being perfectly balanced with respect to inertia forces and moments, i.e. the rocking moments along the shaft, it is also full balanced with respect to the inertia torques, i.e. the twisting moments about the shaft. This feature makes it as perfectly balanced as the Wankel rotary engine.

[22] Higher order harmonic components can be added to or subtracted from the single-lobe, kidney-shape, cam as shown in Fig. 2.

[23] In the multi-lobe cams, of the prior art, the time for one rotation of the multi-lobe cam is longer than the time for a reciprocation of the reciprocating member, thus balance web on them provide no good. Hence, if something makes them in the future desirable, the balance of the engine will necessitate additional counterweight shafts faster than the drive shaft.

[24] Multi-lobe cams impact, as many times as the number of lobes, stronger momentary torques from combustion and even worse torque impacts from inertia, which means as many times stronger impacts for the whole mechanism, gearing included.

[25] Fig. 1 illustrates a way to derive a reciprocation by forcing a pin or a pair of pins to ride on a cam, in this case to produce a strictly sinusoidal reciprocation, called harmonic reciprocation. As it is geometrically shown in Fig. 24, if the pin is kept in permanent contact with the cam, and the center of the pin can move along a line then, as the cam rotates about its axis the pin will perform a harmonic reciprocation.

[26] Fig. 2 depicts the necessary modification of the cam profile in order to add or subtract some higher order Fourier components to the displacement of the pin of the Fig. 1. Here a third order sinusoidal has been added and has been subtracted respectively to derive the other two profiles. The general way for the geometrical derivation of the appropriate cam profile is outlined in Fig. 24.

[27] Fig. 3 shows the way a single-lobe cam, which derives a harmonic reciprocation, needs to be modified if a different motion for the pin is, for some reason, more desirable than the harmonic motion. In most of the drawings, however, the design keeps on for cams deriving a harmonic reciprocation.

[28] Fig. 4 does compare the true dimensions, for the same stroke of the pin, of a three-

lobe and a five-lobe cam, i.e. for identical harmonic reciprocation amplitude, e.g. piston stroke. Also Fig. 4 does compare the dimensions of the three-lobe cam with the dimensions of the single-lobe cam for the same stroke, again, to make it clear that only with a single-lobe cam reasonable dimensions are possible for specific stroke, either for the coaxial cams of Fig. 5 or for simply parallel cams or for the rest ways, here presented, deriving scissors-like action.

[29] Fig. 5 shows one way to force the pin of Fig. 1 to keep contact with the cam frontal surface, let it be called control surface, by virtue of the cooperation of the cam of Fig. 1 with another coaxial cam, the problem lies with the driving of the second shaft, it takes at least five gears to accomplish the differential.

[30] Fig. 6 shows a second way to force the pin to ride on the cam control surface of Fig. 1, i.e. by means of a second cam which is simply parallel, but not coaxial, to the cam of Fig 1.

[31] Fig. 7 proves that the pattern of Fig. 6 easily apply to a single piston.

[32] Fig. 8 illustrates a three cylinder twin-shaft engine. The cams are single-lobe and their counter-rotation takes place by means of a pair of gears (13) and (14).

[33] Fig. 9 is a cross section of the engine of Fig. 8 to reveal a piston arrangement.

[34] Fig. 10 is another cross section of the engine of Fig. 8 and a disassembly, to show the piston rod with its rollers

[35] Fig. 11 is a bottom view of the engine of Fig. 8 to show the two cams as they cooperate, the one being formed as a double-disk to allow a gap for the other cam to pass through, in order to keep them close to make the engine compact and the piston assembly strong, and the balance webs on the two shafts.

[36] Fig. 12 is a transparent view of the engine of Fig. 8 to show the three pairs of cams and the position of the respective pistons.

[37] Fig. 13, 14 and 15 depict a double-head piston four-cylinder or H-4 arrangement.

[38] Fig. 16 shows how a shuttle can be formed in a piston to house the cam throughout its rotation, i.e. to connect the reciprocating member with the rotating member desmodromicly, thereby provide an engine consisting of merely two moving parts, i.e. the rotating component and the reciprocating component.

[39] Fig. 17 shows the displacement of the piston of Fig. 16 as a result of the rotation of the single-lobe cam of Fig. 16. The single-lobe cam has full desmodromic control over the piston, i.e. complete control without any involvement of additional restoring means.

[40] Fig. 18 shows another way to force the pin of Fig. 1 to keep in contact with the control surface of the cam of Fig. 1, i.e. by virtue of wall means, or rails, etc.

[41] Fig. 19 shows the desmodromic control of a reciprocating roller trapped between a pair of non concentric cams and a pair of walls.

[42] Fig. 20 shows the wall version for a double-head piston and a X-type engine at right angle.

[43] Fig. 21 shows the mechanism at 12 successive angles of shaft rotation. The thrust rollers are coaxial to the rollers rolling on the frontal surface of the rotating cam-lobe. There are walls, not shown, where the thrust rollers roll on.

[44] Fig 22 shows an in line three cylinder engine having a single shaft with single-lobe cams, as well as the necessary immovable walls and the various parts disassembled.

[45] Fig 23 shows a shaft with a cam lobe and a piston assembly, with the piston at an offset from the axis of the shaft, as well as immovable walls for taking the thrust loads.

[46] Fig 24 and 25 show the geometrical construction of the cam lobe profile.

[47] Fig 26 shows two counter rotating cam lobes and immovable walls. The piston has a unique pin with rollers.

[48] Fig 27 is the mechanism of Fig 26 with the cam lobes rotating at the same direction.

[49] Fig 28 shows another realization of the mechanism with a unique cam lobe, immovable walls and the piston assembly.

[50] Fig 29 and 30 show a straight four balanced engine with a single shaft.

[51] Fig 31 shows the contact angle between roller and cam lobe. All curves are for the same stroke and for sinusoidal reciprocation, i.e. harmonic. Increasing the size of the single lobe cam, shown at bottom right, it results the basic curve shown at top right, having weaker thrust loads and larger size. The three lobe curve, of similar external size, imparts heavier thrust loads.

[52] Fig 32 to 37 show a desmodromic valve control system.

[53] Fig 38 to 45 analyze the geometry of the mechanism.

[54] Fig 46 shows a mechanism similar to the one shown in Fig 28 to 30 with the difference that the piston has a connecting rod and the thrust rollers roll along paths which are rotatable for a few degrees about the main shaft, providing variable compression.

[55] Fig 47 to 49 show a mechanism based on a camshaft having two different complementary external cams. Each reciprocating roller cam follower rides on the external surface of its own cam.

[56] In the embodiment of Fig. 8 the two shafts counter-rotate by virtue of the equal gears (13) and (14).

[57] The cams on the shaft (7) are made as double-disk cam (9) and (10), to allow the cams (11) of the shaft (8) to pass through, so that no twisting moment is imparted to the piston assembly.

[58] The camlobes (9) and (10) of the shaft (7) and the camlobe (11) of the shaft (8) rotate. The rollers (5), (6) and (4) on the piston assembly (1) rolls along the peri phery

of the camlobes, making the piston to reciprocate inside its cylinder. The proper selection of the profiles of the camlobes and of the diameter and arrangement of the rollers on the piston assembly, make the mechanism full desmodromic, as all rollers are kept permanently in contact to the camlobes.

[59] Adjusting means, such as bolting, springs etc, known in the art, may be added to the rod assembly to provide the desirable clearances or preloading between the roller and the lobe.

[60] Unlike a multi-lobe cam, the rotation of the single-lobe cam is of the same order, i.e. frequency, as the reciprocation of the piston, thereby the webs (12) on the counter-rotating shafts suffice for the full balance of the forces and moments and, as the total kinetic energy of the three harmonically reciprocating members remain constant all along a revolution, there is no inertia torque altogether. The engine of Fig. 8 is as perfectly balanced as a rotary engine, e.g. Wankel rotary engine. The counterweight (12) are also to reduce the main bearing loads.

[61] Fig. 28, 29 and 30 illustrate an even simpler embodiment. Here a single cam (9) in cooperation with a wall or rail (16), as a second rolling surface, completes the scissors. The rail has the advantage of been easily adjustable.

[62] The even firing, straight four engine of Fig 29 has a single, one piece, shaft with a single-lobe cam for each cylinder. The engine is perfectly balanced as regards inertia forces and inertia moments. The rods connecting the upper part of the piston assembly to the lower part of the piston assembly could be just wires, as they are loaded with only tension loads.

[63] Replacing the second shaft (8) of Fig 8 with a wall, as shown in Fig. 22, the result is one shaft and one gearing less but also one first order balance shaft lacking. Now the perfectly balanced three-in-line of Fig. 8 is no longer perfect unless an extra balance shaft is added for the elimination of the rocking moments along the shaft, but with respect to inertia forces and inertia twisting moments, i.e. torques, it remains balanced. A correct clearance between the thrust rollers and the wall, permits the use of the same thrust rollers to roll along the left wall surface as long as the thrust load is to the left direction, and to the right wall when the thrust load is to the right direction. As the direction of the thrust load changes at top and bottom of the reciprocation, where the thrust rollers stop rotating, the transition from the one side wall to the other is smooth and friction free.

[64] As shown from Fig 31, the thrust loads resulting from the mechanism are significantly stronger compared to the thrust loads of the conventional crank-rod mechanism. In order to make the mechanism efficient and reliable, these strong thrust loads must be carried without losing excessive energy in friction. The secondary rotating cam lobe geared to the primary cam lobes, or the immovable wall surfaces,

allows to bear the thrust loads with rollers rolling on them.

[65] Fig 24 shows the way to create a cam lobe profile, given the amplitude of the sinusoidal, i.e. harmonic, reciprocation. The basic curve, upper left, has an eccentricity described as:

[66] E(f)=a + r*sin(f).

[67] A roller having its center on the periphery of the basic curve moves around the curve. Taking a circular disk, like the one shown in upped middle, and subtracting the roller as it rotates around the basic curve periphery, it results the upper right curve and finally the low middle curve. Holding two rollers, like the one used to subtract material from the circular disk, in a distance 2*a from center to center, shown in low middle, and permitting them to move only perpendicularly, the rotation of the cam lobe causes a harmonic reciprocation, along perpendicular axis, of the two rollers assembly, keeping both of them in permanent contact to the cam lobe. In the low right side is shown a groove made in similar way. Using a pair of counter rotating grooves, coaxial or parallel, it can result a reciprocation free from thrust loads.

[68] If the desirable reciprocation is not harmonic, the formula becomes:

[69] E(f)=a + Y(f), where Y(f) is the desirable displacement along the perpendicular axis, relatively to the rotation angle. of the cam lobe.

[70] In Fig 25 the center of rotation of the cam lobe is offset from the axis of reciprocation of the rollers. For harmonic reciprocation the eccentricity of the basic curve, left, becomes:

[71] E(f+f1)=square root ((a + r * sin(f))^2 +d^2), with f1=Arctan ((a+r*sin(f))/d), where d is the offset.

[72] Moving a roller, while keeping its center on the basic curve, it results the cam lobe profile, shown at the middle. In this case the two rollers are in constant distance from each other and reciprocate harmonically as the cam lobe rotates, but they are horizontally offset at 2*d. Two 'offset' counter rotating cam lobes are shown in the right side, with a piston assembly keeping all rollers. Again if the harmonic reciprocation is not the desirable one, the formula becomes :

[73] E(f+f1)=square root ((a + Y(f))^2 +d^2), with f1=Arctan ((a+Y(f))/d).

[74] The above geometrical method apply in the same way for multi lobe cams, for instance two lobe, three lobe etc.

[75] As in the prior art, the space beneath the piston remains available for a second chamber which may serve as a compressor for supercharging etc.

[76] In the following it will be proved that the disclosed solution of the problem of desmodromic or positive control of a reciprocating piston comprising a pair of roller cam followers, by a single cam surface rotating once per reciprocation, is not just one solution but the only possible solution, i.e. it is sufficient and necessary.

[77] In the general case, as shown in Fig 39, a uni-lobe cam profile CP rotates about a center O, with f being the rotation angle. One roller cam follower of radius R1 reciprocates with the piston and moves along an axis X1. Another roller cam follower of radius R2 reciprocates with the piston and moves along an axis X2 parallel to X1. The piston is controlled desmodromicly by the cam with the roller cam follower R1 at one direction, and with the roller cam follower R2 at the opposite direction. E1 is the centers curve of R1 roller while E2 is the centers curve of R2 roller.

[78] There are only six cases.

[79] 1st case: X1 and X2 coincide, R1=R2, O on the coinciding axes.

[80] 2nd case: X1 and X2 are not coinciding, R1 equal to R2.

[81] 3rd case: X1 and X2 coincide, R1=R2, O outside coinciding axes.

[82] 4th case: X1 and X2 coincide, R1 not equal R2, O outside coinciding axes.

[83] 5th case: X1 and X2 coincide, R1 not equal R2, O on the coinciding axes.

[84] 6th case: X1 and X2 are not coinciding, R1 and R2 are not equal.

[85] Case 1, Fig 24. For any given piston's position function Y(f), provided that Y(f)+Y(f+π) = constant for every f, there exist a cam profile CP that provides the specific piston motion. To get the CP, the first step is to derive from the Y(f) the relevant centers curve E of eccentricity E(f)=a+Y(f), where a is a constant. Then, the cam profile CP derives from the centers curve E as an offset, by R, curve, as shown in Fig 38 and 24. The constant a can increase if deficiencies, i.e. cutaways, on the final cam profile occur.

[86] Case 2, Fig 40. When the piston is at TDC, the center of R1 is at A1 and the center of R2 is at A2. So the A1 is a point of maximum eccentricity for the centers curve, while the A2 is a point of minimum eccentricity. The circle with center O and radius OA2 intersects the X1, at the side of A1, at a point B1 which is necessarily a BDC for the piston. The circle with center O and radius OA1 intersects the X2, at the side of A2, at a point B2. Due to desmodromic control the lengths of A1B1 and A2B2 must be equal. So the two triangles OA1B1 and OA2B2 are equal, having all their sides equal. So the O is necessarily in equal distances from X1 and X2 axes, as shown in Fig 25. For any given piston's position function Y(f), provided that Y(f) -Y(f+ π) = constant for any f, there exist a cam profile CP that provides the specific piston motion. To get the CP, the first step is to derive from the Y(f) the relevant centers curve E of eccentricity

[87] E(f+f1)=square root ((a + r * sin(f))^2 +d^2), with

[88] f1=Arctan ((a+r*sin(f))/d), and the a being a constant. Then, the cam profile CP derives from the centers curve E as an offset by R curve, as shown in Fig 38. The constant a can be increased, if necessary, to correct deficiencies on the final cam profile.

[89] Case 3, Fig 41. The offset position of the cam leads to a series of maximum eccentricity points having minimum eccentricity points between them. When the piston is at TDC, the center of R1 is at A1 and the center of R2 is at B1. A circle with center O and radius OA1 intersects the reciprocation axis, at the side of B1, at a point A2. The angle between OA1 and OB1 is φ, with $\varphi < \pi$. For each point of minimum eccentricity on the cam surface, there are two maximums at + φ and - φ angles, and for each point of maximum eccentricity on the cam surface there are two minimums at + φ and - φ, giving infinite maximums and minimums for random φ as shown in Fig 42. In the best case there is a series of a maximum, a minimum, a second maximum and a second minimum at $\pi/2$ angle from each other. This means that the cam profile, if it exists, cannot be a uni-lobe cam profile.

[90] Case 4, Fig 43. When the piston is at TDC, the center of R1 is at A1 and the center of R2 is at B1. When the piston is at BDC the center of R1 is at A2 and the center of R2 is at B2. The angle between OA1 and OB1 is φ 1 and the angle between OA2 and OB2 is φ 2 with φ 1< π and φ 2< π. For each maximum on the cam there are two minimums at angle - φ 1 and + φ 2, and for each minimum on the cam there are two maximums at angles - φ 2 and + φ 1. As in case 3 this necessarily leads to multi-lobe cams, so there is no uni-lobe cam for such a case.

[91] Case 5, Fig 44. The E1 centers curve is the offset by R1 of the cam profile CP and the E2 centers curve is the offset by R2 of the CP. So between E1 and E2 there is a band of constant perpendicular, to the curves, width R1-R2. A random line from the O intersects the E1 at A and B points, it also intersects the E2 at C and D points. The constant distance between the centers of R1 and R2 gives AD=CB, and so AC and DB must be equal. Rotating the AB line for an infinite angle df, the A comes to A1, the B to B1 and so on. AD is constant so A1D1=AD, so AA2=D2D1, with A2 being the point on OA with OA2=OA1, and D2 being the point on OB with OD2=OD1. But A1A2=df*OA1 and D1D2=df*OD1. And because tan(A2AA1)=A1A2/AA2 and tan(D2DD1)=D1D2/DD2, so tan(A2AA1)/tan(D2DD1)=OA1/OD1, so the angle at the longer eccentricity is bigger. But (R1-R2)=AC*sin(A2AA1)=BD*sin(D2DD1), due to the constant perpendicular width between E1 and E2, and AC=BD, so the angles A2AA1 and D2DD1 are equal. Not possible.

[92] Case 6, Fig 45. A1 is the center of R1 at TDC and A2 is the center of the R2 at TDC. B1 is the center of R1 at BDC and B2 is the center of R2 at BDC. Either the φ 1 is not equal π or the φ 2 is not equal π, the cam has multiple lobes, so a uni-lobe cam cannot exist, as in cases 3 and 4. If both φ 1 and φ 2 are equal to π, because of the equal lengths of the A1B1 and A2B2, the O is at equal distances from X1 and X2, which gives OA1=OB2. But OA1=OC1+R1, and OB2=OD2+R2, with OC1=OD2, because A1 and B2 corespond to maximum eccentricity on the cam lobe. So it is

necessary R1=R2. So for not concentric axes the only solution to achieving the desmodromic control is to have roller cam followers of equal radius.

[93] Decreasing until zero the offset d of case 2, it results the case 1, with d=0 and $f1=\pi/2$. So the problem of the positive or desmodromic control of a reciprocating piston, having a pair of roller cam followers, by one only uni-lobe cam surface is solved, the solution provided is the only possible while the only limitation is the piston's position function Y(f) to obey in the rule $Y(f)+Y(f+\pi)=constant$. The practical application of the method is clear: in the general case, given the piston's position function Y(f) the centers curve E(f) is calculated according the formulas given and then the center of the cutting tool of a milling machine follows the specified centers curve creating the cam.

[94] If the roller, as it moves contacting the cam lobe, is held parallel to itself, then instead of the rolling, a sliding takes place. However, only a small part of the periphery of the roller comes in contact to the cam and the rest periphery of the roller, being free, is not necessary. By machining a pair of cams, one external and one internal, as Fig. 32 to 37 show, and holding between them a slim cam follower resulting as the section or the subtraction of the two rollers used initially to configure the two cams, according the geometrical method provided, a desmodromic mechanism of small dimensions, reduced inertia, broad contact surfaces, of fewer components and free of springs, capable of controlling the motion of a reciprocating valve, is provided. Fig 34 shows seven successive positions of the cam follower sliding along the groove. Fig 36 and 37 show the application on a pair of valves. The two cam followers are formed at the two sides of a flat valve holder, whose thrusting surface is not shown.

[95] The present invention is, of course, in no way restricted to the specific disclosure of the specification and drawings, but also encompasses any modifications within the scope of the appended claims.